German Utility Model
GM 78 03 641 U1

Four-Stroke Internal Combustion Engine

Claims

1. A four-stroke internal combustion engine with several cylinders, the inlet ducts of which are directly in fluid communication with one another, and with a cam shaft, the cams of which control the inlet and outlet valves, characterized in that the cam profile (base) (18) produces premature opening of the inlet valve (8) in a known manner, such that exhaust and residual gases pass into the induction system and moreover brings about a late closure of the inlet valve such that with low numbers of revolutions, gas mixture escapes into the induction system and a blocking valve or similar which only opens with a flow directed towards the inlet valve is disposed in the inlet manifold (22) near to each inlet valve.

2. The four-stroke internal combustion engine according to Claim 1, characterized in that the blocking valve in the inlet manifold (22) is provided directly at the attachment point (26) of the same onto the cylinder head (5).

3. The four-stroke internal combustion engine according to Claims 1 and 2, characterized in that the blocking valve is in the form of a leaf spring tongue [or flap] (30).

4. The four-stroke internal combustion engine according to Claim 3, characterized in that the leaf spring tongue [or flap] (30) is disposed on a part (32) provided with an opening (34) which extends at an acute angle to the cross-sectional surface of the inlet manifold (22).

The invention relates to a four-stroke internal combustion engine with several cylinders, the inlet ducts of which are directly in fluid communication with one another, and with a cam shaft, the cams of which control the opening timings of the inlet and outlet valves.

With multi-cylinder four-stroke engines, it is known for the inlet ducts to be connected to one another by a common inlet manifold if the engines are equipped with one or two carburators for all cylinders. The disadvantage of these engines is that due to the gas exchange control system which consists of the cam shaft and the inlet and outlet valves controlled by the latter, it is not possible to eliminate the negative influence of the exhaust and residual gases of a cylinder upon the adjacent cylinders during gas exchange. This is particularly the case when, due to a cam profile which offers a somewhat larger valve opening cross-section during the opening period, an increase in torque and output is to be achieved with approximately the same nominal rotation speed.

With a valve opening cross-section curve with which the closure period is lagging, i.e. with which the valve closes later, an increase in output with higher nominal rotation speeds is achieved by the fact that with higher mixture speeds, the recharge effect occurs.

With low numbers of revolutions however, the momentum of mixture column is too small so that due to the strong induction process in the adjacent cylinder and due to the longer piston path of travel towards top dead center, the cylinder charge is partly emptied, and this leads to a reduction of the torque in the range of up to approximately 3,500 or 4,000 rpm.

With conventional four-stroke internal combustion engines with several cylinders for normal road motor vehicles, one could therefore not take advantage of achieving an increase in torque in the lower revolutions range and an increase in performance with higher nominal rotation speeds due to a special cam profile because the average driver of a motor vehicle can not be expected to put up with the disadvantages resulting from this. These disadvantages are

that with greater valve opening overlaps, exhaust and residual gases pass into the induction system, due to which very uneven idling of the engine with misfiring occurs. The aforementioned reduction of the torque in the lower revolution range with longer opening of the inlet valve as a result of the increase in performance with higher nominal rotation speeds can not be easily accepted either. With racing engines, these disadvantages can of course be tolerated.

It is thus the object of the present invention to avoid these disadvantages, i.e. to provide a four-stroke internal combustion engine for conventional passenger vehicles, and moreover of course for all road motor vehicles, which produce a greater torque and better performance with the same nominal rotation speed without the occupants having to forgo comfort when the engine is idling or a good torque in the lower range of revolutions.

This object is achieved according to the invention in that the cam profile (base) produces premature opening of the inlet valve in a known manner, such that exhaust and residual gases pass into the induction system and moreover effects late closure of the inlet valve such that with low numbers of revolutions, gas mixture escapes into the induction system and a blocking valve or similar, which only opens with a flow directed towards the inlet valve, is disposed in the inlet manifold near to each inlet valve.

DT-PS 472 992 discloses a compound internal combustion engine comprising a charge compressor, internal combustion engine and exhaust gas turbine to dispose a closure component closing towards the charge compressor, by means of which higher exhaust or inlet pressure is prevented from flowing into the charge compressor with at the same time opening of the inlet and outlet components. Here, the closure component is effective dependent upon the pressure of the exhaust gases, the valve opening timings remaining constant.

Altering the valve opening timings for the purpose of achieving an improved level of efficiency over the whole range of revolutions is not intended here.

In order to have as little as possible exhaust and residual gas and non-combusted mixture escape from the cylinder, according to a further feature of the invention, the blocking valve [or flap] in the inlet manifold is provided directly at the attachment point of the same onto the cylinder head. It could, however, also be positioned in the cylinder head itself if the necessary space is available in the inlet duct in front of the inlet valve.

An important prerequisite is of course that the blocking valve [or flap] is not sluggish because it must respond immediately to the gas flow and to the smallest difference in pressure on its two sides. The blocking valve is therefore in the form of a leaf spring tongue [or flap] according to a further feature of the invention. This leaf spring tongue [or flap] is advantageously not attached directly in the inlet manifold, but instead is advantageously disposed on a part which is provided with one or more openings and extends at an acute angle to the cross-sectional surface of the inlet pipe.

The invention is described in greater detail referring to an embodiment.

In the drawings:

Fig. 1 shows a section through the cylinder head, the inlet manifold and the carburator,

Fig. 2 shows an approximately horizontal section through the inlet manifold,

Fig. 3 shows a gas exchange control diagram with the valve control timings for each cylinder,

Fig. 4 shows the valve opening curves for two cylinders with the curve for the piston speed, and

Figs. 5 to 9 show different valve opening curves.

The inlet duct 6, which leads into the combustion chamber 7, is formed in the cylinder head 5 of Fig. 1. The inlet valve 8, which is acted upon by the valve spring 10, is influenced by the rocker lever 12 as regards opening. The rocker lever 12 in turn, which is mounted on the rocker lever mounting 14, is connected to the cam 18 of the cam shaft 20 by a tappet 16.

The inlet manifold 22 adjoins the cylinder head 5 and forms the continuation of the inlet duct 6 up to the carburator 24. The attachment point for the inlet manifold 22 onto the cylinder head 2 is indicated by 26. It can be seen in Fig. 2 that the inlet manifold 22 for each of the four cylinders 1, 2, 3 and 4 forms the associated inlet duct 6 which, as can be seen, are connected to one another below the carburator. The inlet valves are indicated in Fig. 2 for all four cylinders as in Fig. 1 by 8.

The cam 18 for each of the inlet valves 8 has the profile or base which, according to the invention, provides better charging of the combustion chamber than does the conventionally used form which is shown by a dot dashed line and indicated by 17. The cam profile 18 produces the valve opening curve according to Figs. 8 and 9 which will, however, be discussed in greater detail below.

In the inlet duct 6 of the inlet manifold 22, near to its attachment point 26 onto the cylinder head 5, a blocking valve in the form of a leaf spring tongue [or flap] 30 is disposed which only allows a flow of gas in the direction going towards the inlet valve 8. The leaf spring tongue [or flap] 30, which has a very small mass, is attached to a part 32 and lies against this such that it extends at an angle to the cross-sectional plane of the inlet duct 6. The part 32 has a large opening 34 by means of which the part 32 only forms an edge for supporting the leaf spring tongue [or flap].

The blocking valve 30 makes it possible to use cam profiles which have a substantially larger base in comparison to the currently conventional design. In

this way, the volumetric efficiency over the whole range of revolutions is generally increased. During the induction process, in the region A between the blocking valve 30 and the inlet valve, negative pressure prevails which causes a movement of the mixture column in the direction of the inlet valve. Hence, the blocking valve 30 opens [and] if, due to the influences of long inlet valve opening times and gas pulsations, the direction of the mixture flow is reversed, the blocking valve [or flap] 30 closes because it is biased and because negative pressure prevails in space B. It is important to keep the space between the inlet valve 8 and the blocking valve 30 as small as possible so as to only allow a minimal quantity of cylinder charge or residual exhaust gases to flow out. It is shown in Fig. 2 with a dot-dashed line 23 how residual exhaust gases can flow from cylinder 1 into cylinder 3 if no blocking valve 30 is provided.

The control timings for the valves of the individual cylinders are shown by Fig. 3 which are plotted relative to one another corresponding to the ignition sequence 1, 3, 4, 2. The dashed lines 38 show the opening timings of the outlet valves and the full lines 40 show those of the inlet valves. The overlap c of the opening timings of the inlet and outlet valves can be seen here. The dashed vertical lines represent the top dead center (OT) and the bottom dead center (UT) for the respective cylinders.

Fig. 4 shows the valve lift for the inlet and outlet valves of cylinders 1 and 3. The dashed curves 42 apply for the outlet valves, and the curves 44 shown by full lines for the inlet valves. The inlet valve opens approximately 35° before top dead center, and this corresponds to path [or portion] *a*. If with the cylinder 3, i.e. the lower curves 42 and 44 in Fig. 4, one observes the induction process, it can be seen that as the piston speed increases, the valve opening cross-sections also increase such that after 30° piston movement (line A) the piston speed has achieved approximately half of the maximum value, whereas the valve opening cross-section is approximately 30 % of the maximum value. The further ratios with the cylinder 3 are shown by lines B and C. The curve for the piston speed is shown by dots and dashes, and indicated by 46. The path [or

portion] resulting from line OT (top dead center) to line C forms the zone of influence of the adjacent cylinder.

In UT (bottom dead center) the piston speed equals 0. After this, the movement picks up again in the direction of the top dead center OT. This means that the piston partially pushes the sucked-in cylinder charge back out again. The longer the inlet valve remains open, the greater of course is the piston movement in the direction towards OT (top dead center). The piston speed is at almost its maximum value at the point where the inlet valve closes. This is shown by the lower part of the curve 46.

One generally strives to design the valve opening cross-sections to be as large as possible in order to reduce the resistances in the valve opening cross-sections as far as possible. Because, however, the "breathing" system in combustion engines is a swinging system, the opening and closing timings with the inlet valves have to be chosen such that the maximum mixture charge remains in the cylinder. The blocking valve 30 shown in Fig. 1 prevents the mixture charge from flowing out of the cylinder, even if the control timings are chosen by the profile of the cam 18 such that this would occur without the blocking valve.

Figs. 5 to 9 show the control timings possible for the outlet and inlet valve.

With the valve opening cross-section curves 42 and 44 shown in Fig. 5 for the outlet and inlet valve, a compromise is made between the good torque in the range of up to 4000 rpm and a good performance with a nominal rotation speed of approx. 5000 rpm. The overlap phase is indicated by *c*.

According to Fig. 6, by means of a cam profile, which produces the curve 44a, a somewhat larger valve opening cross-section is achieved in the opening period which produces an increase in torque and performance with approximately the same nominal rotation speed. The disadvantage of this embodiment without a blocking valve is, as already said, that larger quantities of exhaust gases pass

into the induction system because of the large valve opening cross-sections with the inlet valve during the overlap period because great negative pressure prevails there and leads to misfirings and causes strong engine vibrations during idling.

With a valve opening cross-section curve 44b for the inlet valve in Fig. 7, the increase in performance is achieved with higher nominal rotation speeds because the recharge effect only comes into effect with the greater mixture speeds. With low numbers of revolutions, the momentum of the mixture column is too small such that by means of the strong induction process in the adjacent cylinder and by means of the piston movement towards OT (top dead center), the cylinder charge is emptied in the closure process after passing UT (bottom dead center), and this leads to a reduction of torque in the region up to approximately 3500 – 4000 rpm. The latter is prevented by the blocking valve.

Figs. 8 and 9 show valve opening cross-section curves 44c for the inlet valve with which the changes to the curves 44a and 44b with respect to the curve 44 are made at the same time. These curves 44c produce the advantages already mentioned in connection with a blocking valve.